CFD Analysis of a vertical tube having internal fins for the Natural Convection

A project report submitted in the partial fulfillment of the requirements for the degree of

Bachelor of Technology

(Mechanical Engineering)

By

Shashank Deorah

Roll No. 108ME081

Department of Mechanical Engineering

National Institute of Technology, Rourkela

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Rourkela

CERTIFICATE

This is to certify that the work in this thesis entitled **"CFD analysis of natural convection through vertical tube with internal helical fins"** by **Shashank Deorah**, has been carried out under my supervision in partial fulfillment of the requirements for the degree of **Bachelor of Technology** in **Mechanical Engineering** during session 2011-2012 in the Department of Mechanical Engineering**, National Institute of Technology, Rourkela.**

To the best of my knowledge, this work has not been submitted to any other University/ Institute for the award of any degree or diploma.

Prof. Ashok Ku. Satapathy

(Supervisor)

Associate Professor

Department of Mechanical Engineering

NIT, Rourkela

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CONTENTS

Nomenclature

T_w	Wall temperature
T_∞	Surrounding temperature
k	Thermal conductivity
h_c	Surface heat transfer coefficient
δ_t	Boundary layer thickness
β	Coefficient of thermal expansion
ρ	Density
X	Vertical distance
R	Radial distance
U	Axial velocity of liquid
V	Radial velocity of liquid
g	Acceleration due to gravity
ν	Kinematic viscosity
α	Thermal diffusivity
A_c	Cross sectional area
A_s	Surface area
Q	Rate of heat flux
η_f	Fin efficiency
I_o	Bessel function

ABSTRACT

The heat transfer rate to a fluid flowing in pipe can be enhanced by the use of internal fins. This thesis concerned with computer simulation study of vertical tube with helical fins used to enhance their heat transfer performance subjected to natural convection heat transfer. All the main parameters which can significantly influence the heat transfer performance of finned tube has been analyzed. Natural convection in a vertical tube without fins was taken as the reference tube and different fin patterns such as a single fin with large no. of turns like coiled shape and large no. of fins with single turn is compared with reference tube on the basis of different parameters such as heat transfer rate, surface nusselt number, heat transfer coefficient, fin effectiveness etc. There are some dimensionless numbers which affect the natural convection such as nusselt number which is the function of Reynolds number, grashof number and prandlt number, Rayleigh number which is the product of grashoff and prandtl number. After getting best fin configuration compared it with different fin profile such as rectangular cross section, tapered fin with trapezoidal cross section and hyperbolic cross section. All the computer simulation has been done on the ANSYS 13.0 . The Navier-stokes equations were used to solve for the fluid flow inside the tube and the Boussinesq approximation was used to get the buoyancy effect. Aluminium is used for the fin material and air is taken as the fluid flowing inside the tube and the flow is taken as laminar. It was found that the large number of fins with single turn is more efficient then other fin patterns, as there is less flow resistance, high heat transfer rate.

CHAPTER~1

1. INTRODUCTION

Convection is a process which involves mass movement of fluids. Natural convection occurs due to temperature difference which produces the density difference which results in mass movement, this process is called natural or free convection. For example, assume a plate which is maintained isothermal at temperature T_w and the surrounding temperature is T_∞. On getting heated, the fluid near the wall moves up due to the effect of buoyancy and this hot fluid is replaced by cold fluid moving towards the wall. Hence a circular current is set up due to density difference. There is a boundary layer adjacent to the plate where the velocity and temperature and velocity vary from plate to free stream. Initially the velocity increase with increasing distance from the surface and reaches a maximum and then decrease to approach zero value. This is because of action of viscosity diminishes rapidly with distance from plate, while density difference decreases more slowly.

The used of heat transfer enhancement has become widespread during the last so many years. The need of heat transfer enhancement is to reduce the size and cost of heat exchanger equipment, or increase the heat duty for a give size heat exchanger. This goal can be achieve in two ways active and passive enhancement. The active enhancement is less common because it requires addition of external power (e.g., an electromagnetic field) to cause a desired flow modification. In the passive enhancement, it consists of alteration to the heat transfer surface or incorporation of a device whose presence results in a flow field modification. The most popular enhancement is the fin.

Fins are the extended surfaces which are used to enhance the rate of heat transfer dissipation from heated surfaces to air. Fins can be placed on plane surfaces, tubes, or other geometries. These surfaces have been used to increase heat transfer rate by adding additional surface area and encouraging mixing. When number of fins are used to enhance heat transfer under natural convection conditions the optimum geometry of fins (corresponding to a maximum rate of heat transfer) should be used, provided this is compatible with available space and financial limitations. The common fins used extensively to increase the rates of natural convection heat transfer from systems are rectangular fins because such fins are simple and cheap, to manufacture. The heat transfer to the fluid flowing through a cylindrical pipe by the heat dissipating surfaces can be obtained mainly by using the mechanisms of heat transfer by forced convection, natural convection and by radiation heat transfer. This paper mainly concerned with those issues related to the heat transfer obtained mainly by natural convection.

A great number of experimental and analytical work has been done on vertical and horizontal finned tube subjected to natural convection. Kayansayan [2] studied the thermal characteristic of fin and tube heat exchanger, Rao [3] studied the heat transfer from horizontal fin array, Yang [4] conducted an experiment on mixed convective cooling of a fin in a channel, Sharif and Bergman [5] worked on enhancement of PCM melting in enclosure with horizontal finned internal surface. Myhren [6] worked on improving thermal performance of ventilation radiators using internal fins. Baek [7] studied on heat transfer enhancement using straight and twisted internal fins.

Applications of internal fins

1. Internal fins are used in compact heat exchangers.
2. Internal fins are used in phase change material storages (PCM). PCM are used to balance the temporary temperature alteration and to store energyin several practical fields like automobile industries.
3. The rate of heat transfer from fluid flowing through the microchannels can be greatly enhanced by use of internal fins.
4. Internal fins used in Improving the thermal performance of ventilation radiators.

CHAPTER~2

2. LITERATURE SURVEY

Muñoz et al. [8] done analytical work on internal helically finned tubes for parabolic trough design by CFD tools. The application of finned tubes to the design of parabolic trough collectors has some losses as the pressure losses, thermal losses and thermo-mechanical stress and thermal fatigue. The result shows an improvement potential in parabolic trough solar plants efficiency by the application of internal finned tubes.

SAZALI [9] experimental study of a vertical internally finned tube subjected to natural convection heat transfer. The length of tube was 100mm. the tube taken for the experiment has inner diameter 80mm and the outer diameter 90mm. The tube contains four radial, straight, and equally spaced around the circumference of the tube. Other dimensions like height of the fins are 100mm and the length of the fins are 25mm. Air was used as a working fluid in the experiment. The result shows that the value of Nu for vertical cylinder under variables time varies with the temperature is increasing.

Myhren et al. [6] studied heat output optimization of a ventilation radiator by varying the distribution of vertical, longitudinal convection fins. The investigation was made using Computational Fluid Dynamics simulations while analytical calculations were used for different flow and heat transfer mechanisms. The results showed that heat transfer can be increased in the section where ventilation air is brought into the room by slightly changing the geometry of the fins like decreasing the fin to fin distance. The small change in internal design could mean considerable increase in thermal efficiency for the ventilation radiator as a whole.

Wang et al. [10] studied heat transfer performance of internally finned tubes with blocked core-tube was numerically investigated by the realizable k–e turbulence model with wall function method using FLUENT. By using 3 kinds of lateral fin profiles, S-shape, Z-shape and V-shape, were studied and compared. The corresponding correlations of Nusselt number and friction factor were obtained for different-shape internally finned tubes. The result showed that tubes with S-shape fins and Z-shape fins were best profile as compared with V-shape fins, and moreover, tube with Z-shape fins had the best performance.

Giri et al. [11] worked on the role of natural convection in many applications like ice-storage air-conditioning. A mathematical formulation of natural convection heat and mass transfer over a shrouded vertical fin array is developed. The base plate were kept at a temperature below the dew point of the surrounding moist air due to this, occurrence of condensation of moisture on the base plate, while the fins may be partially or fully wet. The results showed that beyond a certain stream wise distance, further fin length does not improve the sensible and latent heat transfer performance, and that if dry fin analysis is used under moisture condensation conditions, the overall heat transfer will be lowballed by about 50% even at low buoyancy ratios.

Papadopoulous et al. [12] done the Numerical study of laminar fluid flow in a curved elliptic duct with internal fins. The study of the fully developed laminar incompressible flow inside a curved duct of elliptical cross-section with four thin and internal longitudinal fins is done using the improved CVP method. Results showed that the friction factor increases for large fins and for high Dean numbers and in some cases, it has dependent on the cross-sectional aspect ratio. The thermal results show that the heat transfer rate is increased by the internal fins and that it depends on the aspect ratio.

Foong et al. [13] conducted the numerical study to investigate the fluid flow and heat transfer characteristics of a square micro channel with four longitudinal internal fins. 3-D numerical simulations were performed on the micro channel with variable fin height ratio in the presence of a developed laminar flow. Constant heat flux boundary conditions were assumed on the external walls of the square micro channel. Results obtained of the average local Nusselt number distribution along the channel length are as a function of the fin height ratio. The analytical study was carried out for different fin heights and flow parameters.

Aziz [14] et al. measured the heat transfer rate for different fin profile such as rectangular, trapezoidal, and concave parabolic (finite tip thickness). Results obtained from the comparison based on the relationship between the dimensionless heat flux, the fin parameter, and dimensionless tip temperature for all three geometries.

CHAPTER~3

3. THEORY

The science of heat transfer is concerned with the generation, use, exchange, and conversion of heat and thermal energy between physical systems. Heat transfer is the discipline of thermal engineering that concerns the calculation of rate at which heat flows within the medium, across the interface or from one surface to another. There are different modes of heat transfer which includes:

a. Heat transfer through conduction
b. Heat transfer through convection
c. Heat transfer through radiation

3.1 Natural convection

Convective heat transfer is the transfer of heat from one place to another place by the fluid movement. Convection is usually occurred in liquids and gases. The causes of convection can be described as either natural (free) or forced convection. The difference between natural and forced convection is important for heat transfer through convection. The main cause of Natural convection or free convection is due to temperature differences which affect the density and the relative buoyancy of the fluid. Heavier components move down while the lighter components rise which leads to fluid movement. Hence gravitational field plays an important role in natural convection. The examples of natural convection is the rise of smoke from a fire, boiling water in the pot in which the hot and less dense water from the bottom layer moves downward and the cool and high dense water moves upward to the top of the pot. Natural convection will occur due to variation in density between the two fluids, the acceleration due to gravity that drives the convection to a larger distance through the convecting medium.The of convection can be determined by the Rayleigh number (Ra).

Consider a plate maintained isothermal at temperature T_w and the surrounding temperature is T_∞. The fluid near the wall moves up on getting heated because of the effect of buoyancy and this hot fluid is replaced by the cold fluid moving towards the wall. Thus there is a loop creates due to the density difference. There will always be a boundary layer adjacent to the wall, either in the natural convection or forced convection, where the temperature and velocity vary from plate to the free stream.

The thermal boundary layer is considered as the stationary fluid film which is responsible for heat conduction and then heat is transported by fluid motion.

Heat transfer rate from wall to fluid is :

$$Q_c = \frac{-k*A*(T_\infty - T_W)}{\delta_t} \qquad (1)$$

Thermal boundary layer thickness δt is defined as the thickness at which:

$$(T_\infty - T_W) = 0.99\,(T_\infty - T_W)$$

(2)

Let $$\frac{k}{\delta_t} = h_c$$

$$Q_c = h_c * A * (T_W - T_\infty) \qquad (3)$$

The rate of heat transfer Qc increase with increase in value of heat transfer coefficient hc. On increasing the velocity of fluid, the film thickness decreases and value of heat transfer coefficient increases.

3.2 Fluid flow

All the numerical simulation done on the CFD for the fluid flow were based on the Navier-stokes Equation. Donaldson [15] worked on the free convection in the case of vertical cylinder with linear wall temperature gradient. The equations which describe the problem of steady natural convection about a vertical cylinder of radius R with a prescribed wall temperature, are as follows:

Density variation with temperature is:

$$\frac{1}{\rho} = \frac{1}{\rho_o}[1 + \beta(T - T_o)] \tag{4}$$

The equation of continuity, momentum and energy in cylindrical co-ordinates which apply to the fluid flow with boundary conditions are:

Continuity equation: $$\frac{\partial U}{\partial X} + \frac{\partial U}{\partial R} + \frac{U}{R} = 0 \tag{5}$$

Momentum equation: $$\mathrm{U}\frac{\partial \mathrm{U}}{\partial \mathrm{X}} + \mathrm{V}\frac{\partial \mathrm{U}}{\partial \mathrm{R}} = -\mathrm{g} - \frac{1}{\rho}\frac{\partial \mathrm{P}}{\partial \mathrm{X}} + \nu\left(\frac{\partial^2 \mathrm{U}}{\partial \mathrm{R}^2} + \frac{1}{\mathrm{R}}\frac{\partial \mathrm{U}}{\partial \mathrm{R}}\right) \tag{6}$$

Energy equation: $$U\frac{\mathrm{dT}}{\mathrm{dX}} + \mathrm{V}\frac{\partial \mathrm{T}}{\partial \mathrm{R}} = \alpha\left(\frac{\partial^2 \mathrm{T}}{\partial \mathrm{R}^2} + \frac{1}{\mathrm{R}}\frac{\partial \mathrm{T}}{\partial \mathrm{R}}\right) \tag{7}$$

Where X is vertical distance measured upward from origin, R is radial distance, U is the axial velocity upward of the fluid, V is the radial velocity of the fluid.

Boundary conditions are as follows:

$$\begin{aligned} &U=0,\ V=0 &&: \text{ at } R=a, \\ &U=0 &&: \text{ at } X=0, \\ &T=Ts(X) &&: \text{ at } R=a, \\ &T=To &&: \text{ at } X=0, \end{aligned} \tag{8}$$

From boundary conditions, we can see that equation (6) reduce to

$$0 = -g - \frac{1}{\rho}\frac{\partial \rho}{\partial X} + \nu\left(\frac{\partial^2 U}{\partial R^2} + \frac{1}{R}\frac{\partial U}{\partial R}\right) \tag{9}$$

From equation (7) and (9)

$$U\frac{\partial U}{\partial X}+V\frac{\partial U}{\partial R}=g\beta\big(T-Ts(X)\big)+\nu(\frac{\partial^2 U}{\partial R^2}+\frac{1}{R}\frac{\partial U}{\partial R}) \quad (10)$$

3.3 Dimensionless parameters

Reynolds number:

The Reynolds number (Re) [16][17] is a dimensionless number which is defined as the ratio of inertial forces to viscous force.

$$Re=\frac{UL}{\nu}$$

Prandtl number:

Prandtl number is the ratio of kinematic or momentum diffusivity (ν) to the thermal diffusivity (α).

$$Pr=\frac{\nu}{\alpha}$$

Grashoff number:

Grashoff number is the ratio of buoyancy force to the viscous force acting on the fluid.

$$Gr=\frac{g\beta L^3(Tw-T\infty)}{\nu^2}$$

Rayleigh number:

Rayleigh number is the product of grashoff number and the prandtl number. In natural convection Rayleigh number is used instead of grashoff number to correlate heat transfer.

$$Ra = \frac{g\beta L^3(Tw - T\infty)}{\nu\alpha}$$

Nusselt number:

The Nusselt number is defined as the ratio of convective to conductive heat transfer across the boundary.

$$Nu_L = \frac{convective\ heat\ trasfer\ coefficient}{conductive\ heat\ transfer\ coefficient} = \frac{h_c L}{k}$$

In natural convection, the transition from laminar to turbulent flow is determine by critical value of grshoff number.

When, $\frac{\mathrm{Gr}}{\mathrm{Re}^2} \ll 1$, Forced convection

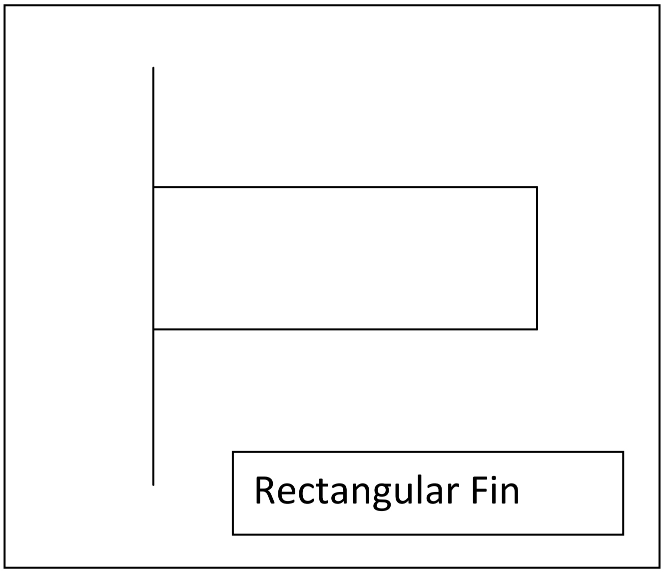
$\frac{\mathrm{Gr}}{\mathrm{Re}^2} \gg 1$, Natural convection

$\frac{\mathrm{Gr}}{\mathrm{Re}^2} \cong 1$, Natural and forced convection are of same order of Magnitude

CHAPTER ~4

4. Theoretical Formulation

4.1 Rectangular fin



[18] The energy balance equation for an element having rectangular fins made of material of uniform thermal conductivity is

The rate of heat conduction into the element = rate of heat conduction out of element + . rate of heat convection from the element surface

The rate of heat conduction in the element is the function of distance x which can be given as

$$Q(x) = -kA_c \frac{dT(x)}{dx}$$

Now, $$Q(x) = Q(x) + \frac{d}{dx}[Q(x)]dx + h_c dA_s[T(x) - T(\infty)]$$

By using element surface area $\mathrm{dA_s} = \mathrm{Pdx}$, we get

$$\frac{\mathrm{d^2T}}{\mathrm{dx^2}} - \frac{\mathrm{h_cP}}{\mathrm{k}}[\mathrm{T(x) - T(\infty)}] = 0 \qquad \text{eq. (11)}$$

Let as assume that

$$\theta(x) = T(x) - T(\infty)$$

$$m^2 = \frac{h_c P}{kA_c}$$

Hence eq. (11) becomes

$$\frac{d^2\theta}{dx^2} - m^2\theta = 0 \qquad \text{eq. (12)}$$

The general solution of this equation is

$$\theta(x) = C_1 e^{-mx} + C_2 e^{mx} \qquad \text{eq. (13)}$$

Let us assume that the fin is of finite length and loss of heat from its tip is convective.

The boundary conditions are:

$$-k[\frac{d\theta}{dx}]_{x=c} = h_c\theta_{x=L}$$

By using this boundary condition and rearranging the equation, we get

$$\frac{\theta(x)}{\theta_0} = \frac{\cosh\{m(L-x)\} + \left(\frac{h_c}{mk}\right)\sinh\{m(L-x)\}}{\cosh(mL) + (\frac{h_c}{mk})\sinh(mL)}$$

And,

$$Q_{fin} = \sqrt{h_c P k A_c}(T_0 - T_\infty)\frac{\sinh mL + (\frac{h}{mk})\cosh mL}{\cosh mL + (\frac{h}{mk})\sinh mL}$$

4.2 Parabolic Fin

[18] Let us consider a fin has length ‘L’, thickness ‘t’ which is at the base and width ‘w’ which is perpendicular to the plane of the paper has base temperature T_w and surrounding temperature T_∞. consider the thickness is the function of x.

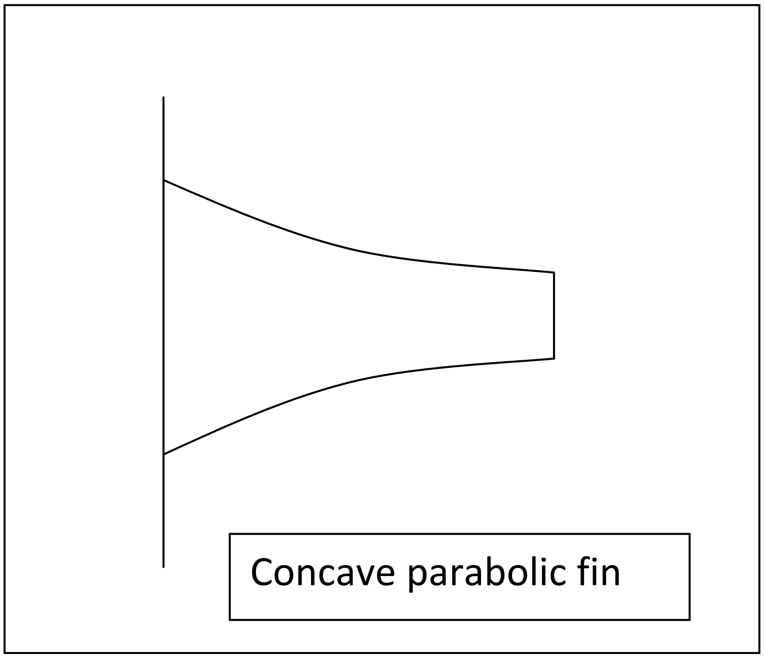
A_s is the surface area and A_c is the cross sectional area.

The eq. of the profile of the fin is

$y = ax^2 + b$

At x=0, y=0

X=L, y=$\frac{t}{2}$



Now, energy balance equation of the element is

$$Q_x = Q_{x+dx} + Q_{convective}$$

$$or, \quad h_c dA_s[T(x) - T(\infty)] = -\frac{d}{dx}[-kA_c\frac{dT}{dx}]dx$$

Now let us surface element area

$$dA_s = Pdx \approx 2wdx$$

$$\theta = T(x) - T(\infty)$$

$$put\ y = \frac{t}{2}(\frac{x}{L})^2$$

$$m^2 = \frac{2h}{kt}$$

$$we\ get,$$

$$x^2\frac{d^2\theta}{dx^2} + 2x - m^2L^2\theta = 0$$

This equation is an eular equation whose solution is obtained by putting $x = e^z$

The heat transfer rate from the parabolic fin is

$$Q_{fin} = \frac{wtk}{L}\left[-\frac{1}{2} + \frac{1}{2}\sqrt{1+4m^2L^2}\right](T_w - T_\infty)$$

$$n_{fin} = \frac{-\frac{1}{2}+\frac{1}{2}\sqrt{1+4m^2L^2}}{m^2L^2}$$

In the designing of parabolic fins the following observations should be noted carefully:

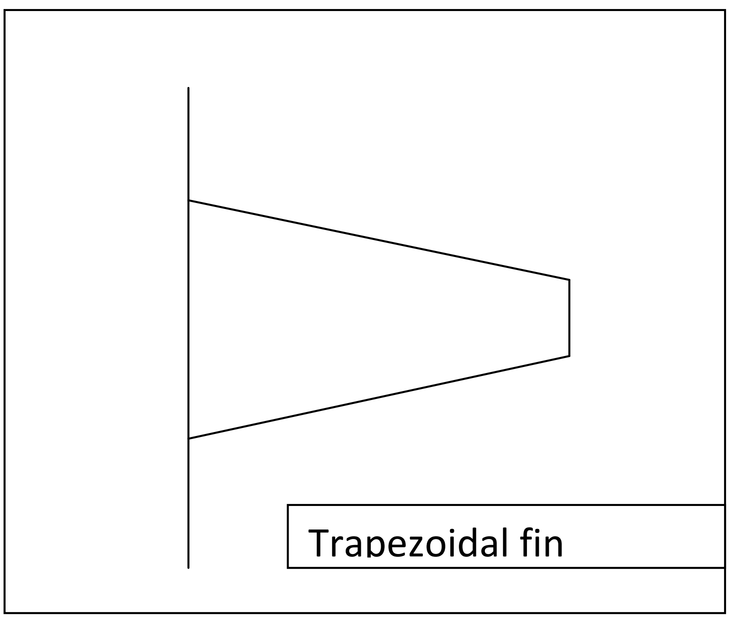
1. The fin must have the minimum material for the most economical fins.
2. The heat flux is independent of fin thickness, and heat conduction rate is independent of distance from the fin base.

4.3 Trapezoidal fin

[19] Consider a trapezoidal fin of cross sectional area A, surface S, length L. For variable fin area energy balance equation is at a distance x from the fin base is given as:

$$\frac{d^2\theta}{dx^2} + \frac{1}{A}\frac{dA}{dx}\frac{d\theta}{dx} - \frac{1}{A}\frac{dS}{dx}\frac{h_c}{k}\theta = 0$$

Where, A is the cross sectional area and S is the surface area of the element.



$$S(x) = 2(x - x_t)\sqrt{1 + (\frac{y-y_t}{x-x_t})}$$

$$\frac{y - y_t}{x - x_t} = \frac{y_b}{L}$$

Where x_t and y_t are the distance from fin base to fin tip in horizontal and vertical directions respectively.

$$hence, \frac{dS}{dx} = \sqrt{1 + (\frac{y_b}{L})^2}$$

now the energy equation becomes

$$\frac{d^2\theta}{dx^2} + \frac{1}{x}\frac{d\theta}{dx} - \frac{P^2}{x}\theta = 0$$

Where, $P = \frac{hL}{ky_b}\sqrt{1 + (\frac{y_b}{L})^2}$

If we take a special case of trapezoidal fin such as triangular fin then the solution of the above equation is

$$\frac{\theta}{\theta_b} = \frac{I_0(2P\sqrt{x})}{I_0(2P\sqrt{L})}$$

Where I_0 is the modified Bessel function of zero order and first kind.

4.4 Fin effectiveness

Fin effectiveness is defined as the ratio of heat transfer rate with fin to the heat transfer rate without fin from the surface. [1]

$$\varepsilon_f = \frac{heat\ transfer\ rate\ with\ fin}{heat\ transfer\ rate\ without\ fin} = \frac{Q_o}{h_c A \theta_o}$$

Where A is the cross sectional area of the fin.

4.5 Fin efficiency

Fin efficiency is the major parameter which is used yo determine the fin performance. It is defined as the: [1]

$$n_f = \frac{actual\ heat\ transfer\ from\ the\ fin}{max.heat\ transfer\ from\ the\ fin\ consider\ that\ entire\ surface\ is\ at\ base\ temperature}$$

$$= \frac{Q_o}{h_c A_f \theta_o}$$

CHAPTER~5

5. Results and Discussion

Analysis has been done on five tubes of same dimensions but having different fin configuration or fin profile with same fin height.

Description of the tubes:

1. Tube1 : This tube is the plane vertical tube without fins.
2. Tube2 : this tube is a vertical tube having a single helical fin of rectangular fin profile with large number of turns along the length.
3. Tube3 : this tube is a vertical tube having ten helical fins of rectangular cross sectional area with single turn along the length.
4. Tube4 :This tube is a vertical tube having ten helical fins of trapezoidal cross sectional area with single turn along the length.
5. Tube5 : this tube is a vertical tube having ten helical fins of concave parabolic cross sectional area with single turn along the length.

Problem data :

Tube no.	Inner Diameter (mm)	Outer Diameter (mm)	Fin Profile	Fin Height (mm)	Fin Thickness (mm)
1.	50	53	Without fin	5	2
2.	50	53	One rectangular fin	5	2
3.	50	53	10 rectangular fin	5	2
4.	50	53	10 trapezoidal fin	5	Varying from 2 to 4 mm
5.	50	53	10 concave parabolic fin	5	Varying from 2 to 4 mm

Table 5.1 Problem data for different fin configuration and fin profile.

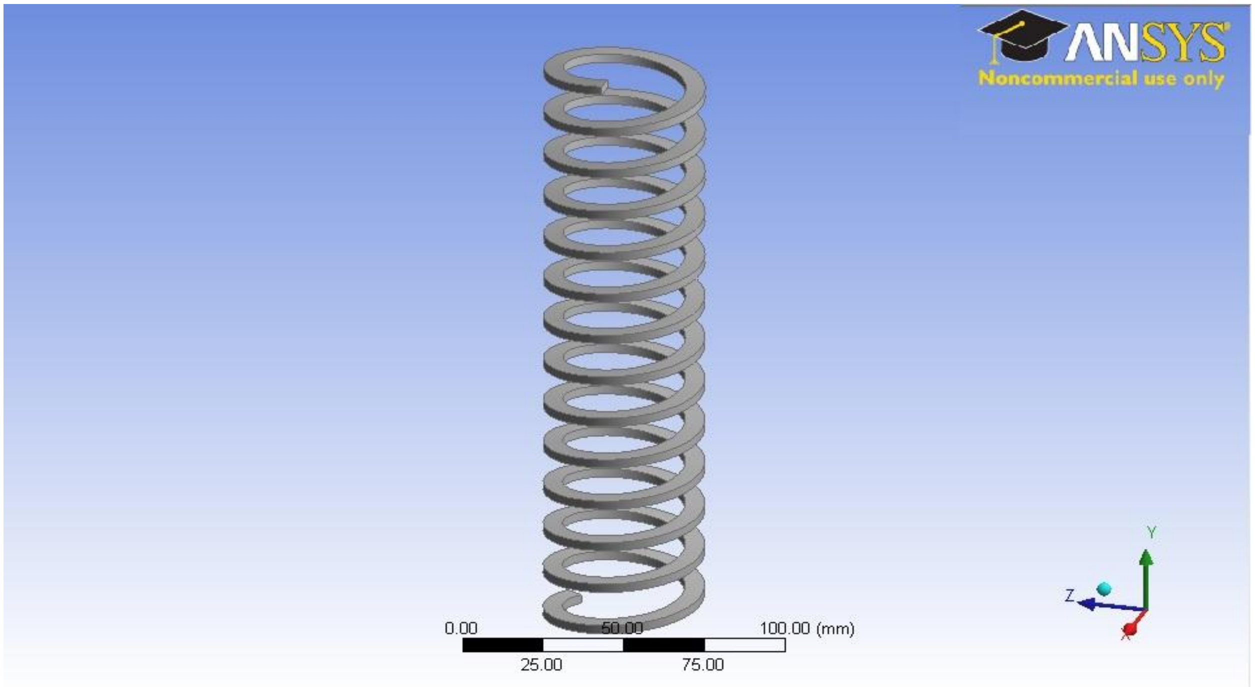


Fig 5.1 One helical fin with large number of turns

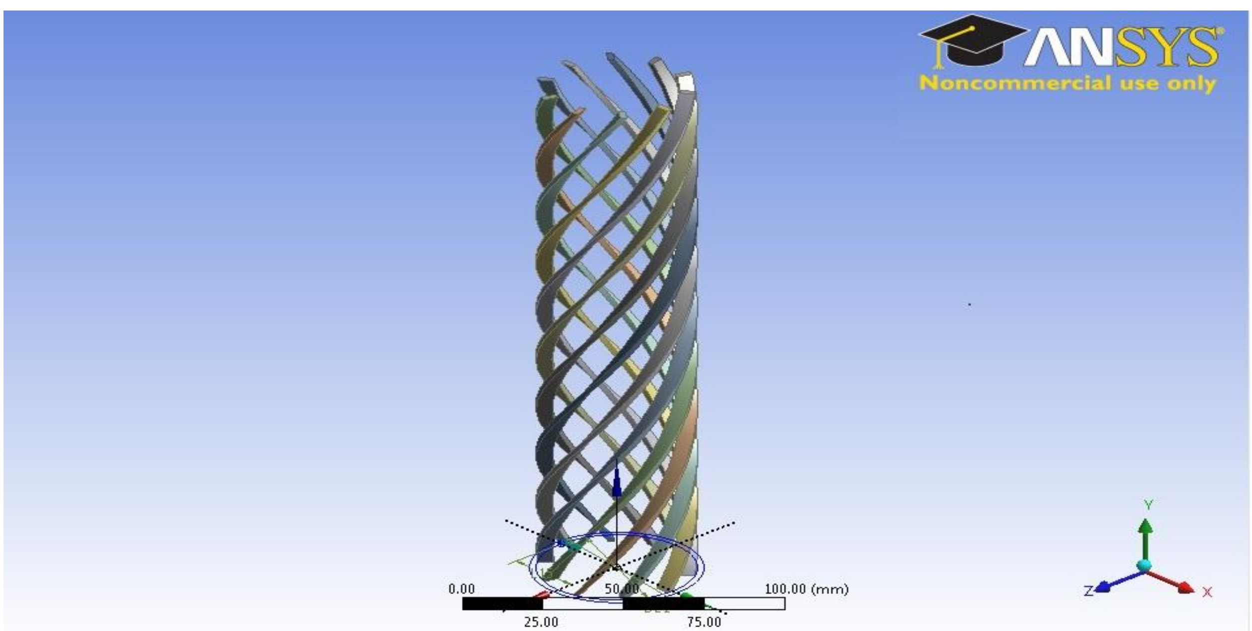


Fig. 5.2 Ten helical fins with single turn

Material Properties

```
  Material: air (fluid)

      Property                        Units      Method       Value(s)
      ---------------------------------------------------------------
      Density                         kg/m3      boussinesq   1.225
      Cp (Specific Heat)              j/kg-k     constant     1006.43
      Thermal Conductivity            w/m-k      constant     0.0242
      Viscosity                       kg/m-s     constant     1.789401e-05
      Molecular Weight                kg/kgmol   constant     28.966
      Thermal Expansion Coefficient   1/k        constant     0.003334
      Speed of Sound                  m/s        none         --

   Material: aluminum (solid)

      Property               Units    Method     Value(s)
      ---------------------------------------------------
      Density                kg/m3    constant   2719
      Cp (Specific Heat)     j/kg-k   constant   871
      Thermal Conductivity   w/m-k    constant   202.4
```

Values of dimensionless number for different tubes:-

Tube	Grashoff no.	Prandtl no.	Rayleigh no.	Flow
1	1.6212×10^6	0.707	1.146×10^6	Laminar
2	9.8955×10^5	0.707	7.86×10^5	Laminar
3	2.4699×10^5	0.707	1.746×10^5	Laminar
4	1.4466×10^5	0.707	1.1138×10^5	Laminar
5	1.0139×10^5	0.707	7.80703×10^6	Laminar

Table 5.2 values of dimensionless number for different tubes.

5.1 Temperature contours for different tubes:

Tube1:

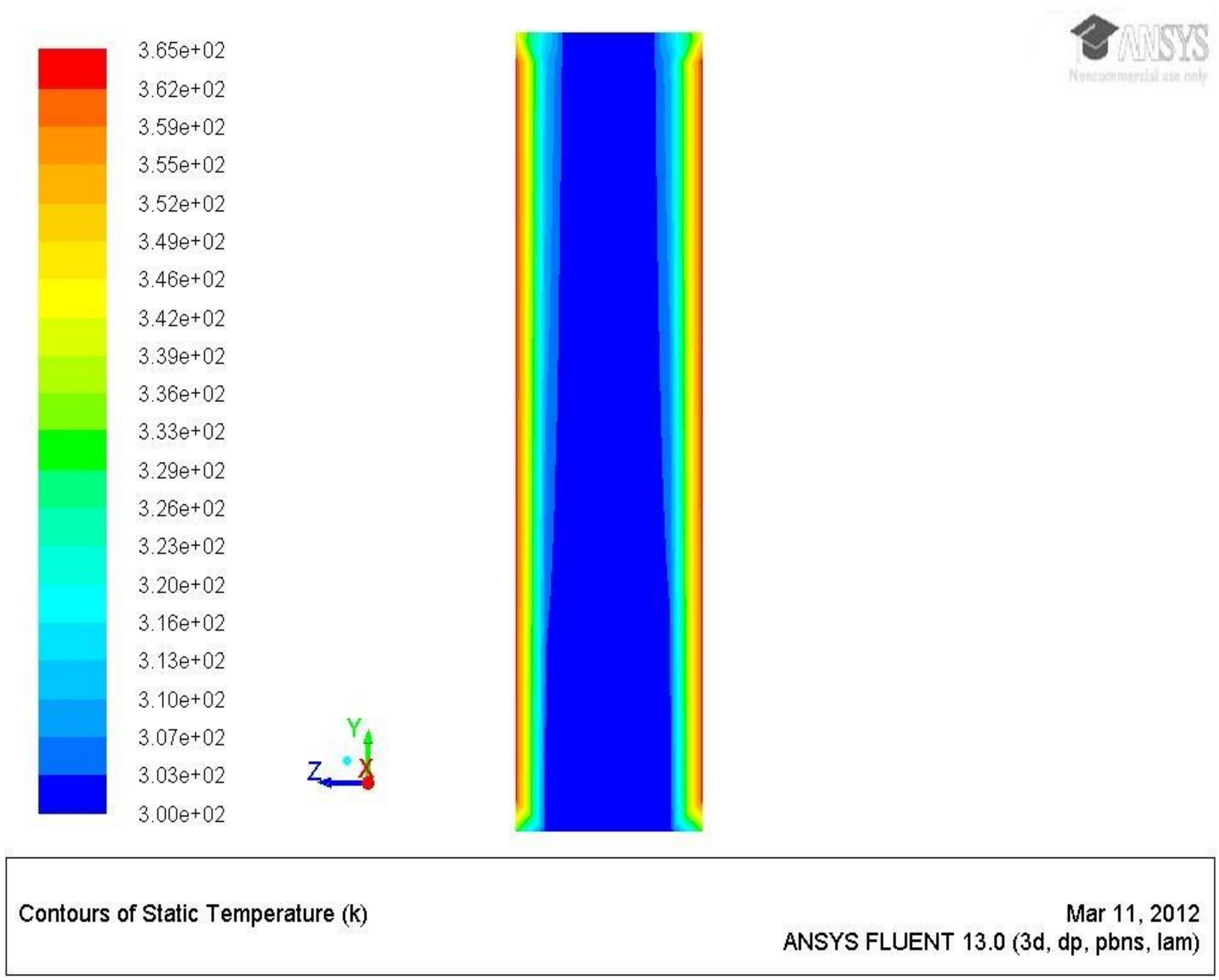


Fig. 5.3 Temp. contour of vertical tube without fin

Tube2:

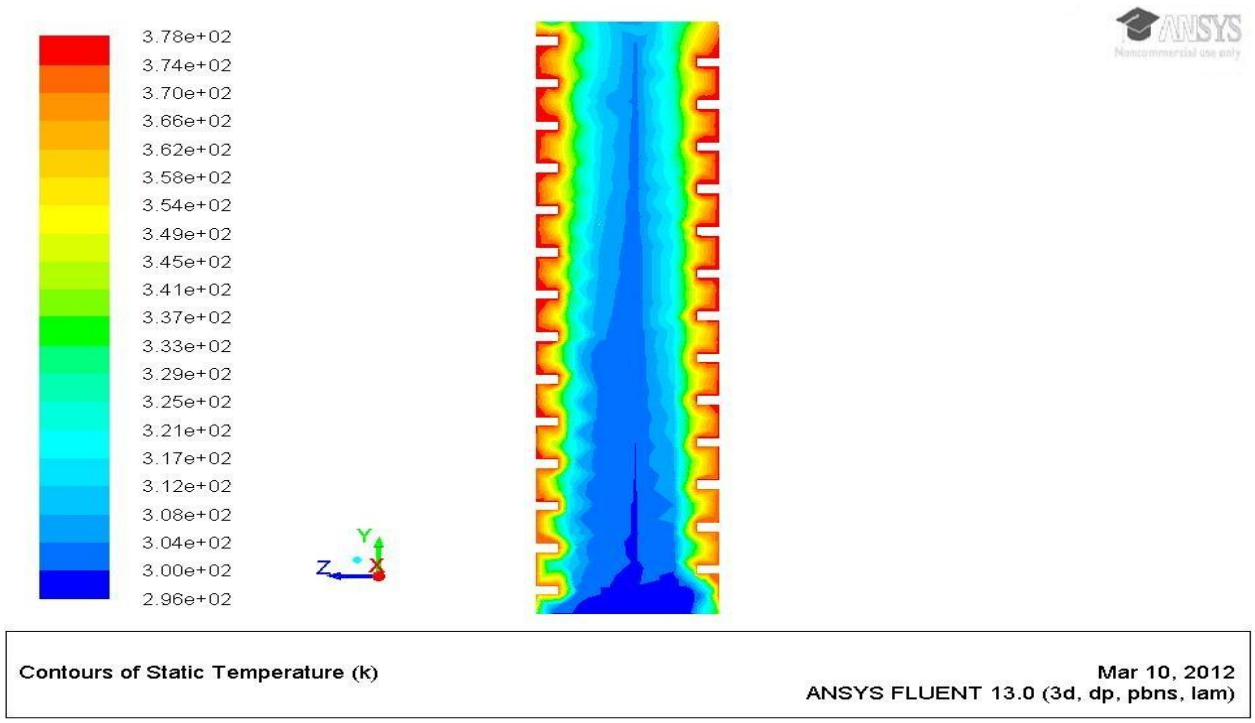


Fig 5.4 Temp. contours of vertical tube with rectangular fin profile

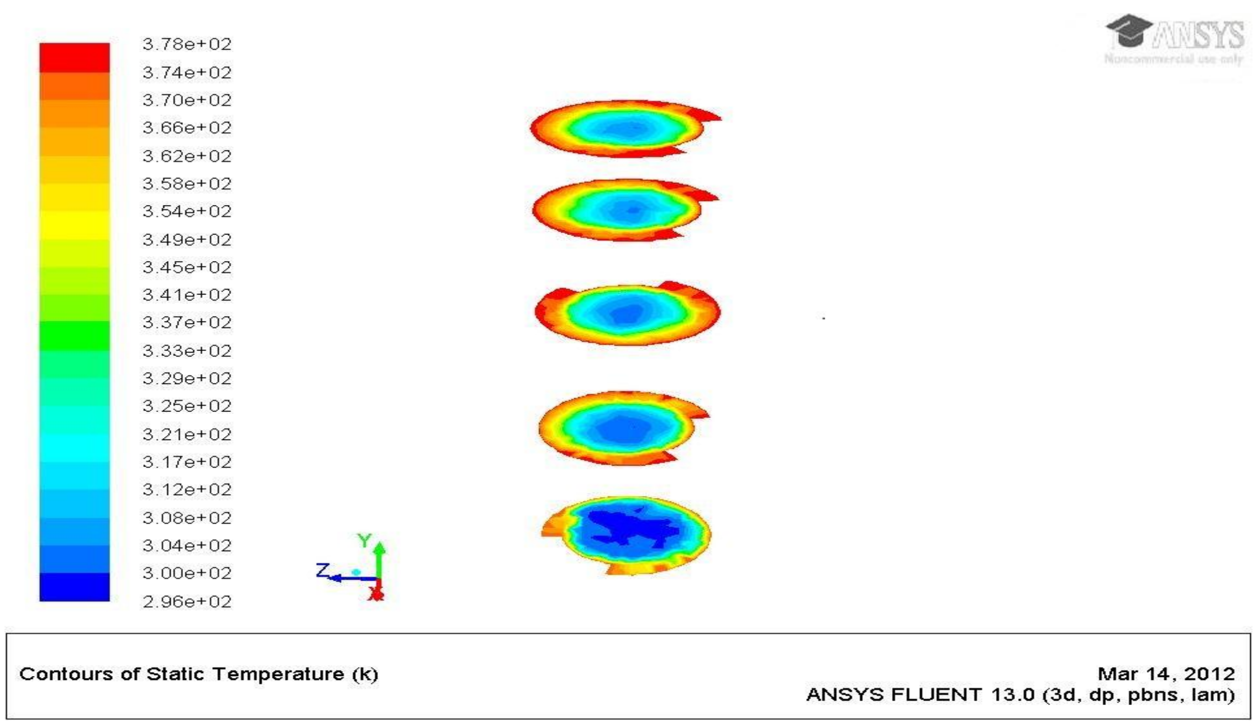


Fig 5.5 Temp. contours of tube2 in different horizontal palne

Tube3:

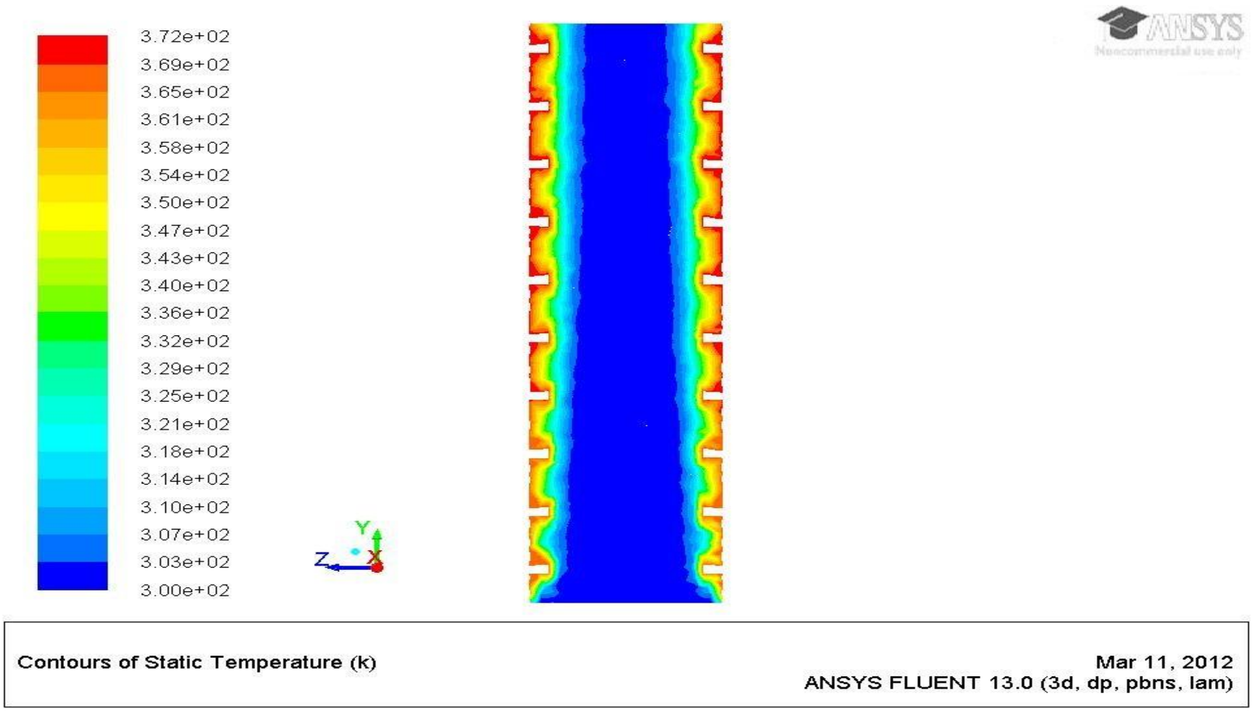


Fig 5.6 Temp. contours of tube3 in vertical plane

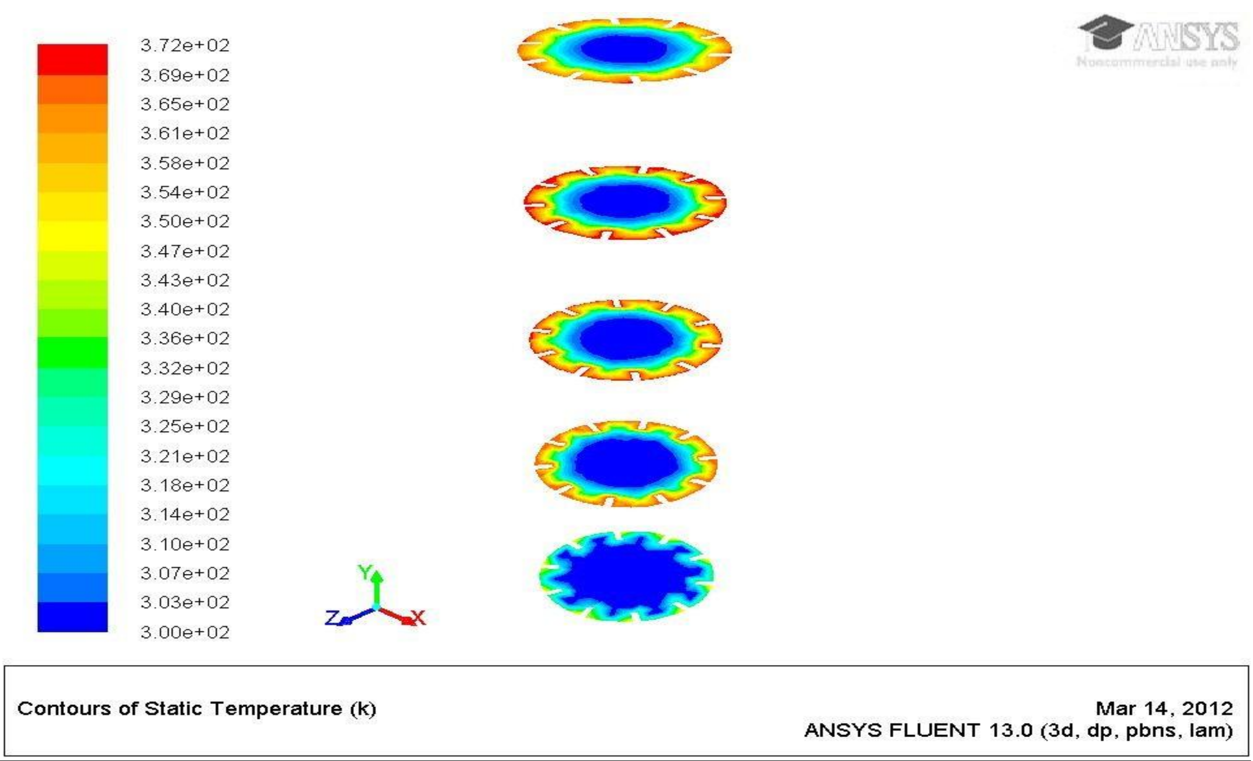


Fig 5.7 Temp. contours of tube 3 in horizontal planes

Tube4 :

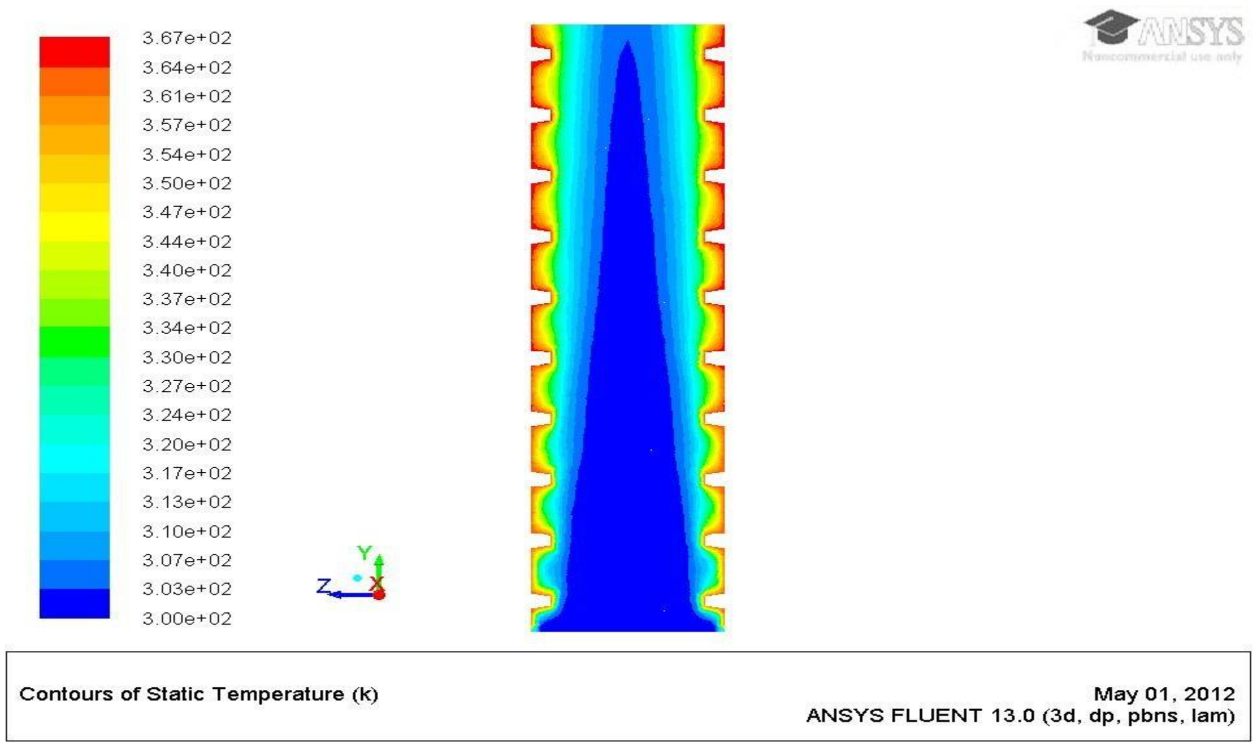


Fig 5.8 Temp. contours of tube 4 in vertical plane

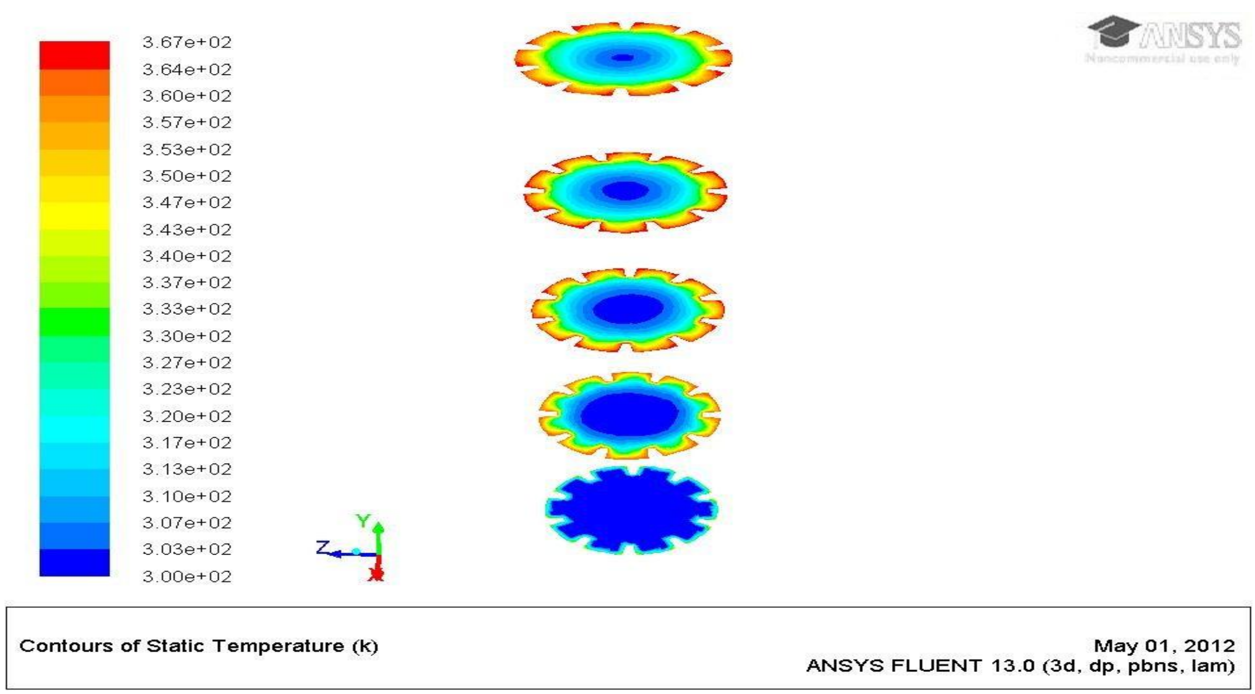


Fig 5.9 Temp. contours of tube4 in horizontal planes

Tube5 :

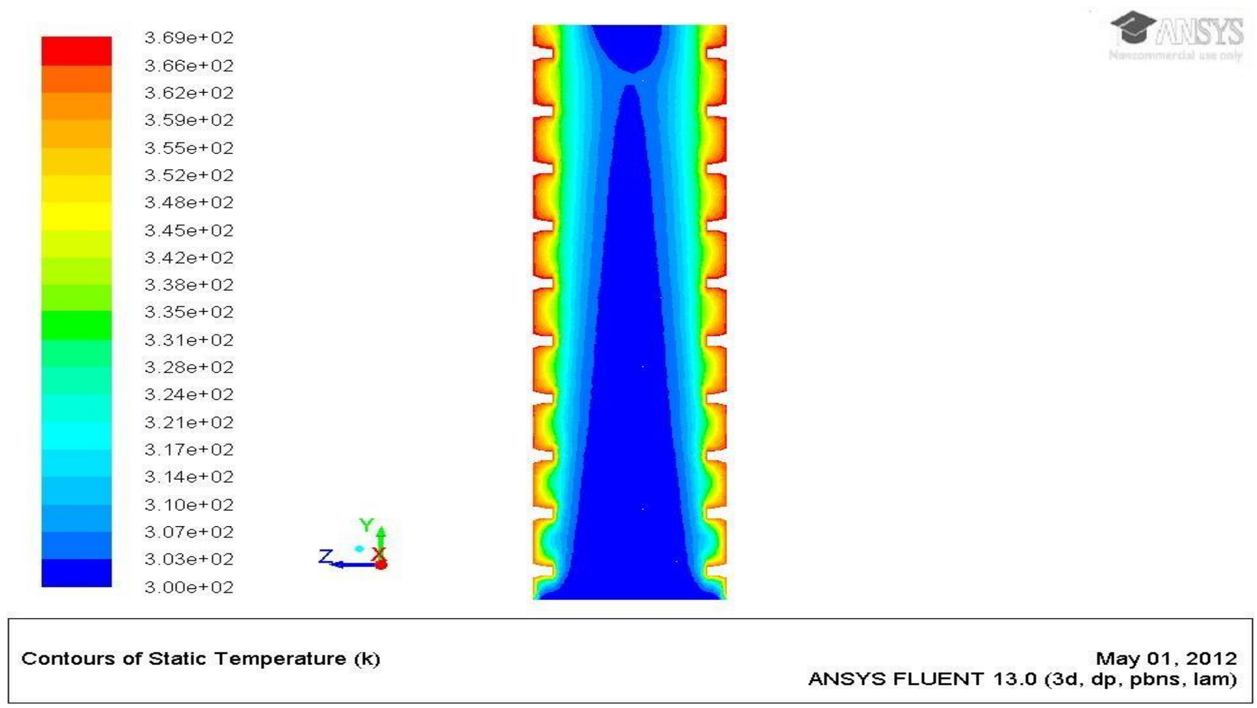


Fig 5.10 Temp. contours of tube5 in vertical plane

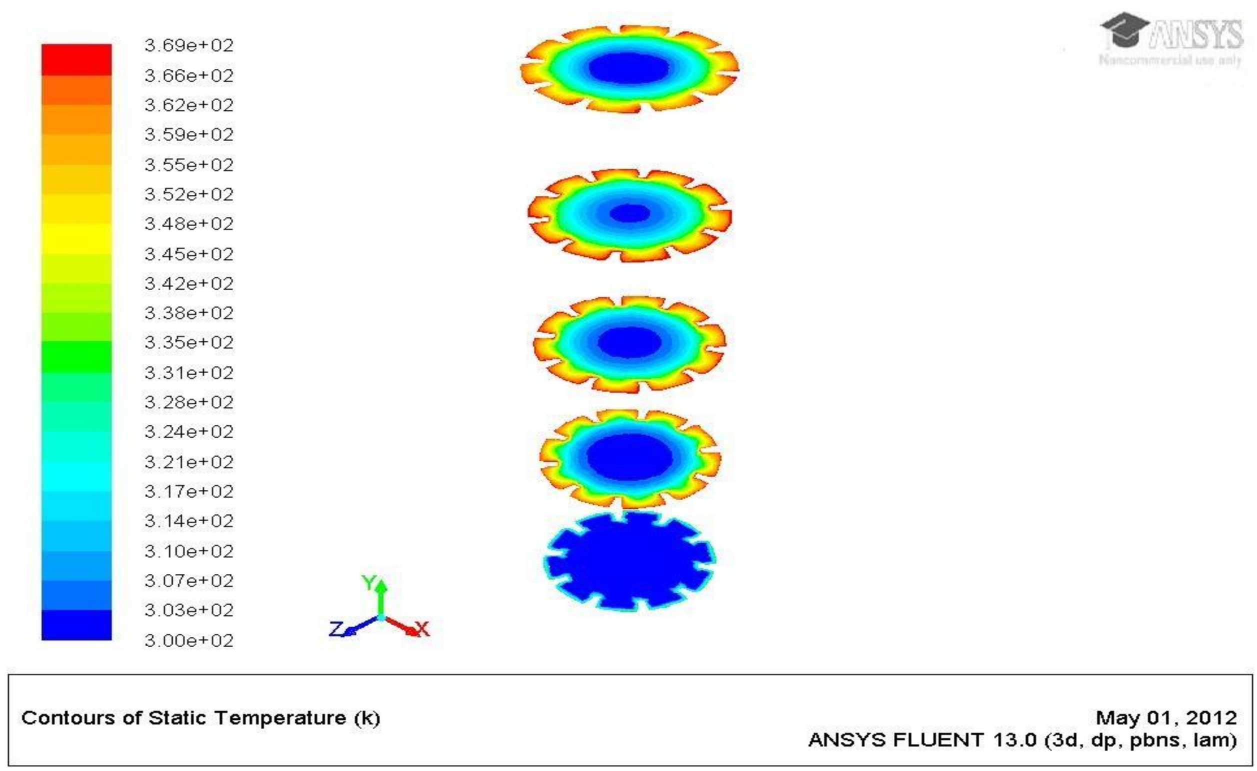


Fig 5.11 Temp. contours of tube5 in horizontal planes

5.2 Surface Nusselt number:

Tube1:

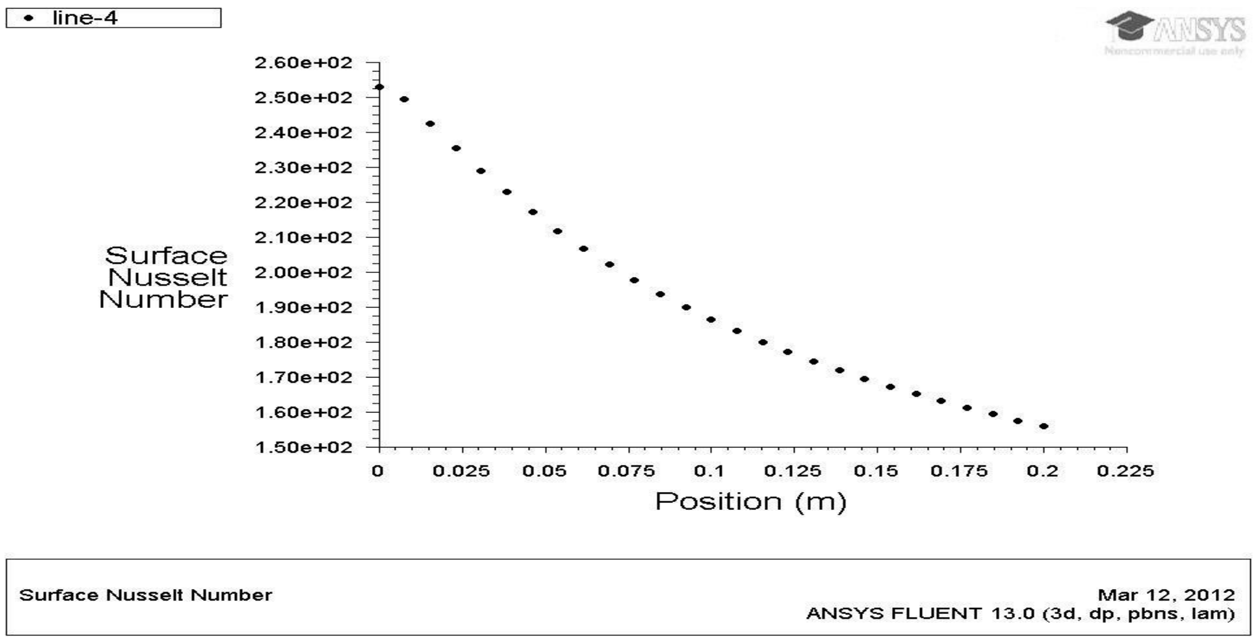


Fig. 5.12 Line-4 passes from the tube surface

Tube2:

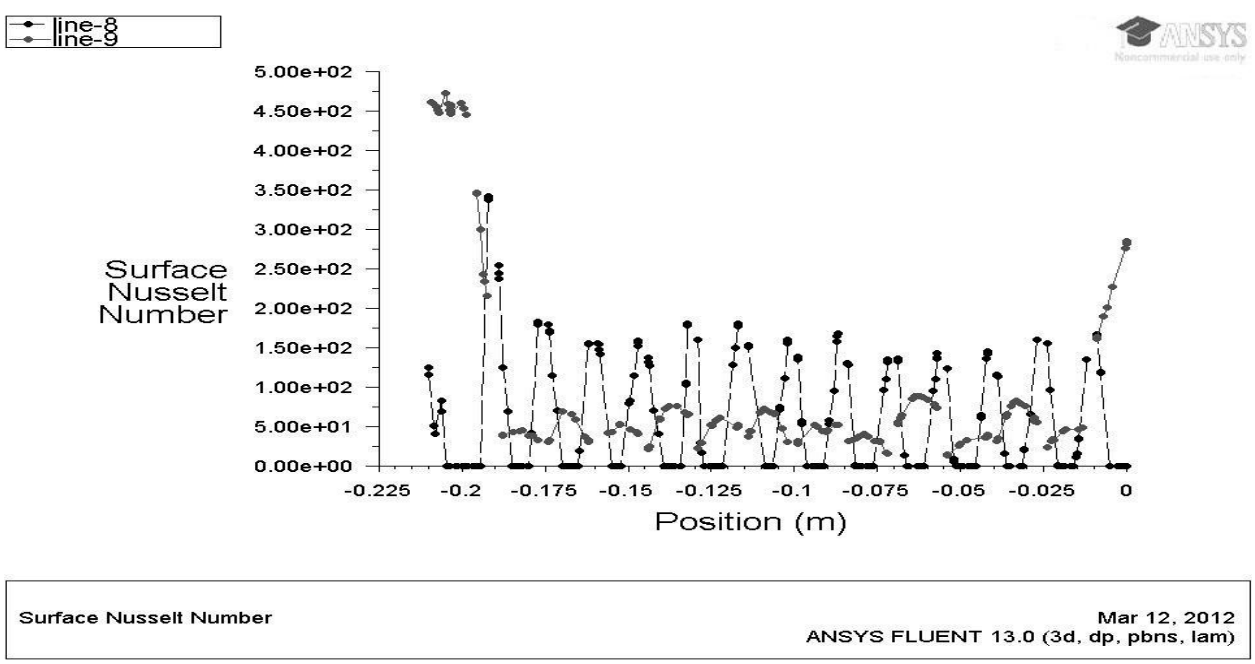


Fig. 5.13 line-8 passes from fin tip and line-9 passes from fin base.

Tube3:

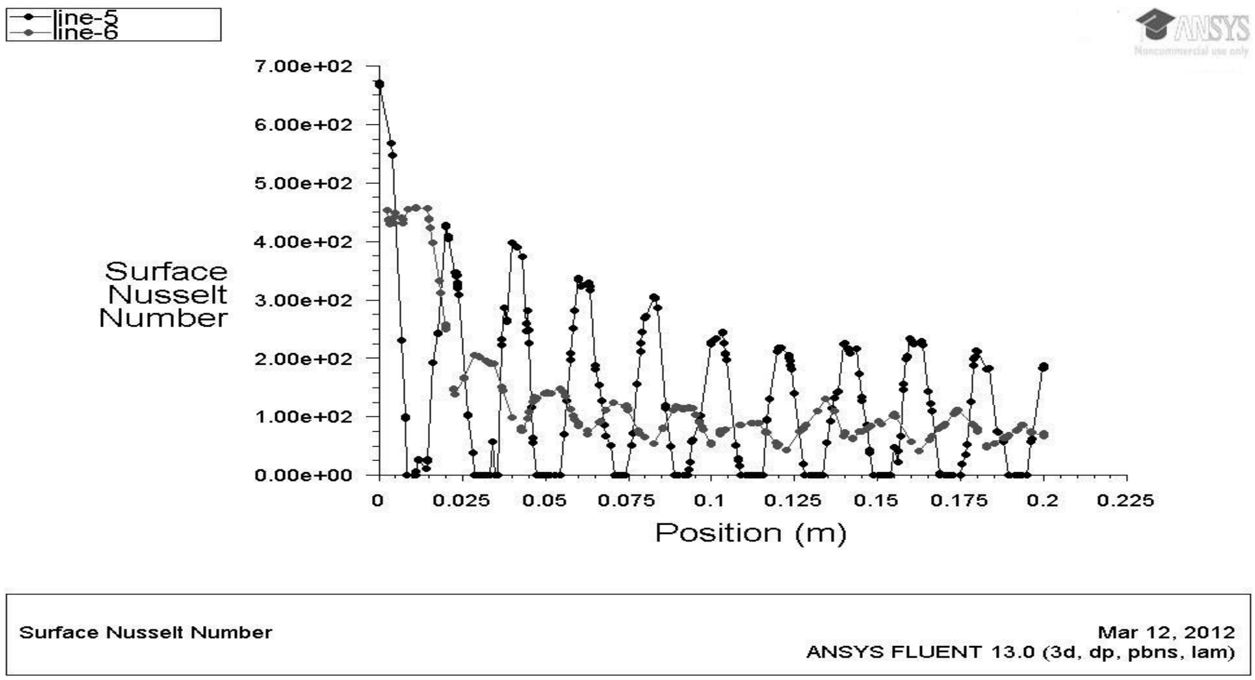


Fig. 5.14 line-5 passes from fin tip and line-6 passes from fin base.

Tube4

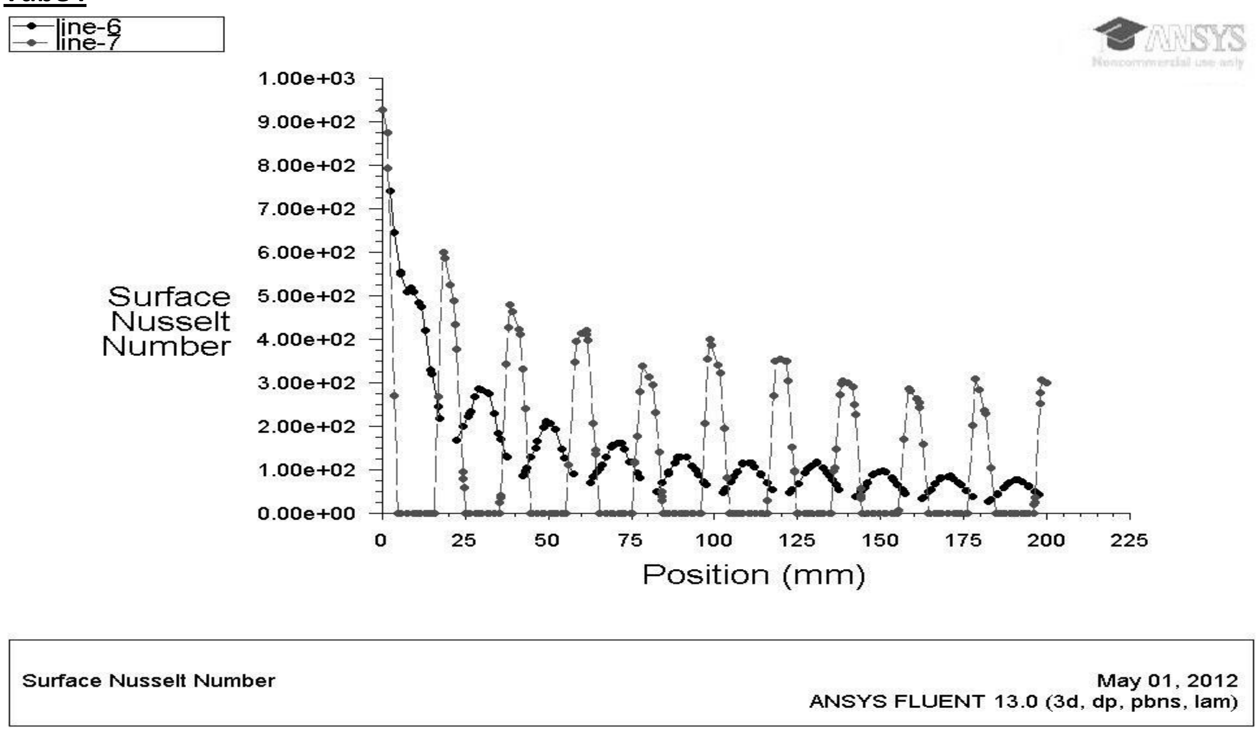


Fig.15 Line-6 passes from fin base and line-7 passes from fin tip.

Tube5 :

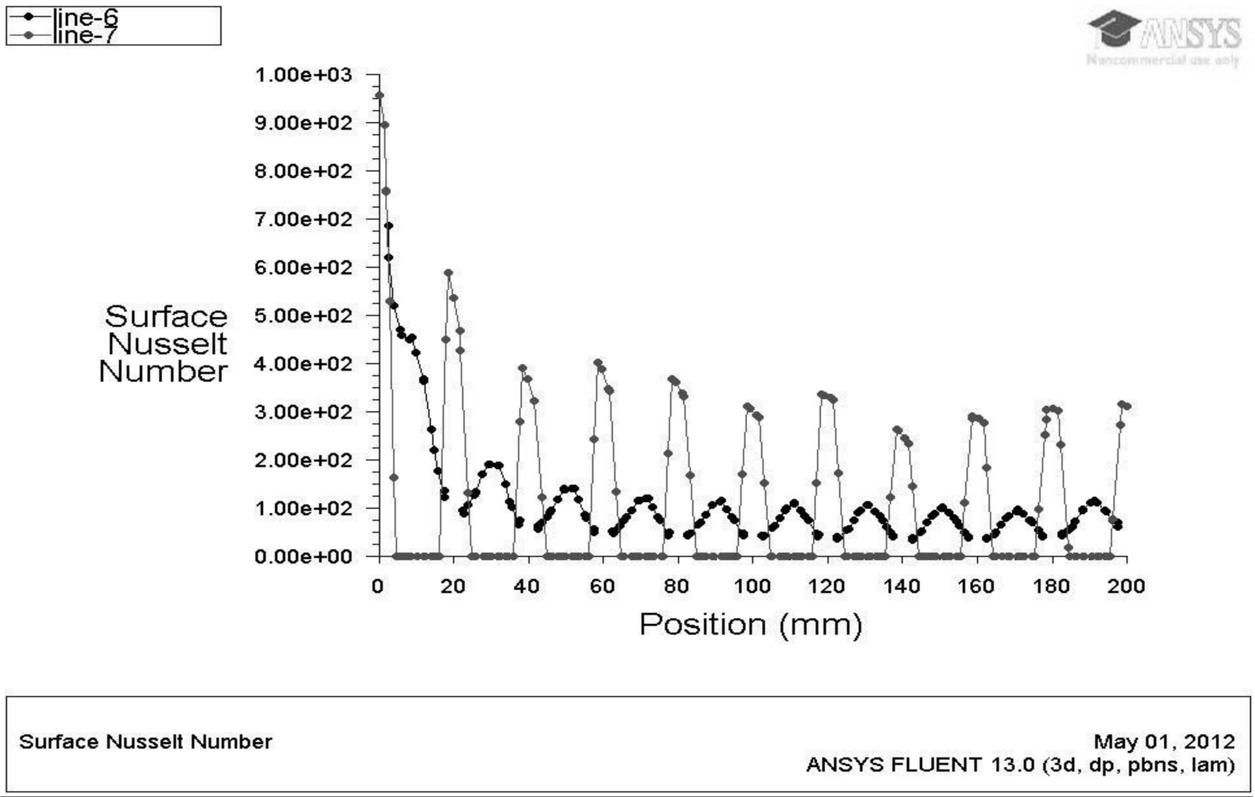


Fig. 5.16 Line-6 passes from fin base and line-7 passes from fin tip.

5.3 Surface heat transfer coefficient

Tube1 :

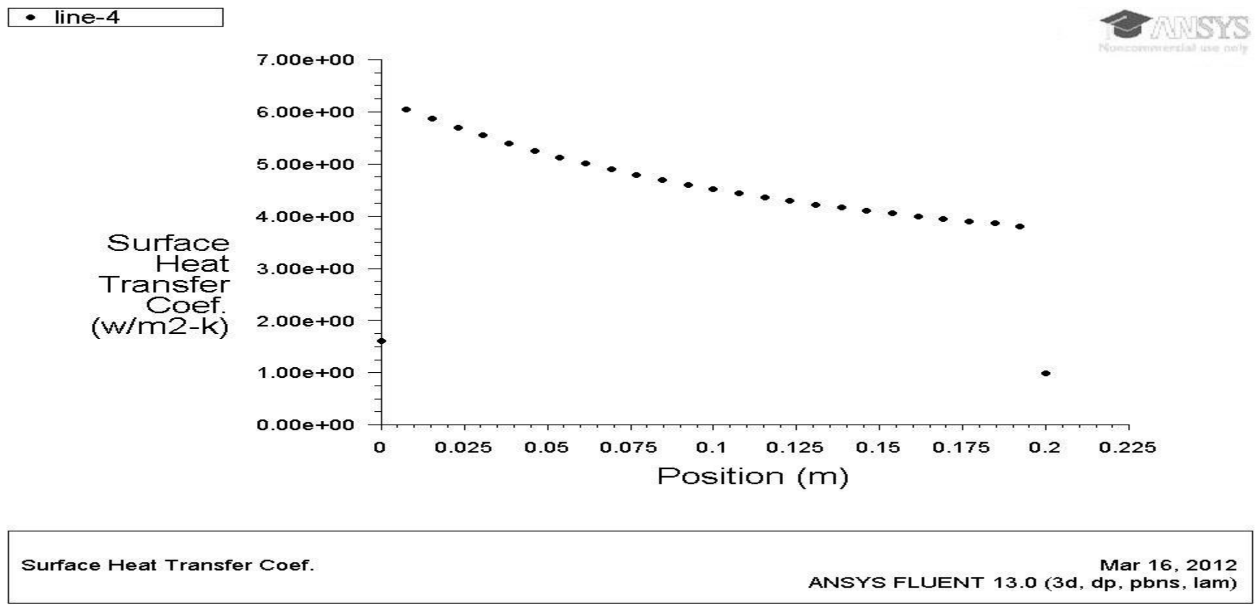


Fig. 5.17 Line-4 passes from tube surface

Tube2 :

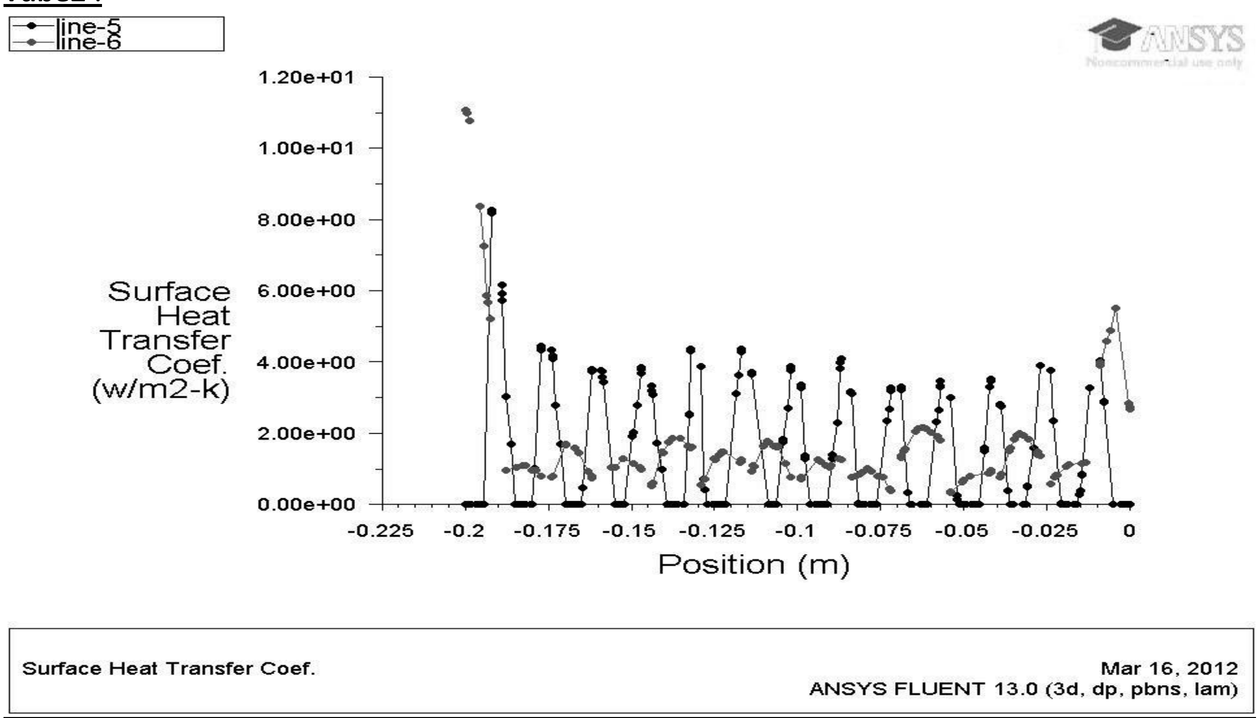


Fig. 5.18 line-5 passes from fin tip and line-6 passes from fin base.

Tube3 :

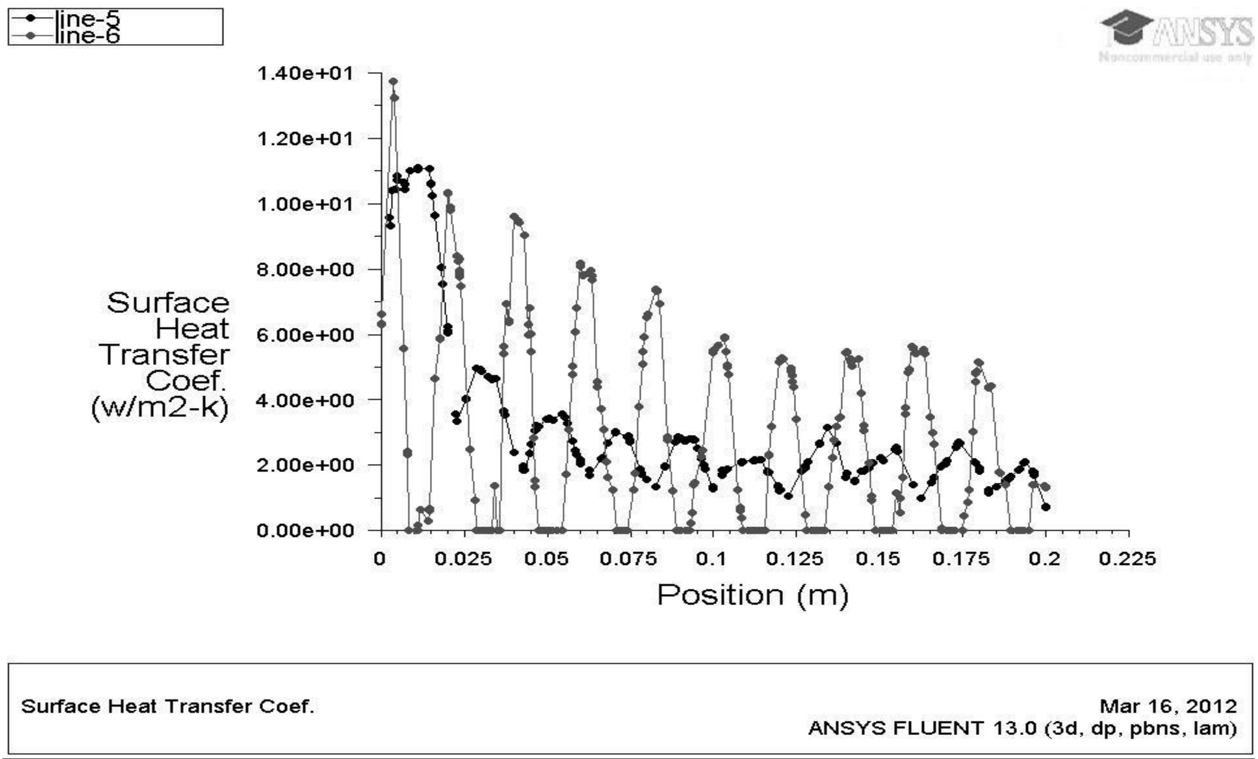


Fig. 5.19 line-5 passes from fin base and line-6 passes from fin tip.

Tube4 :

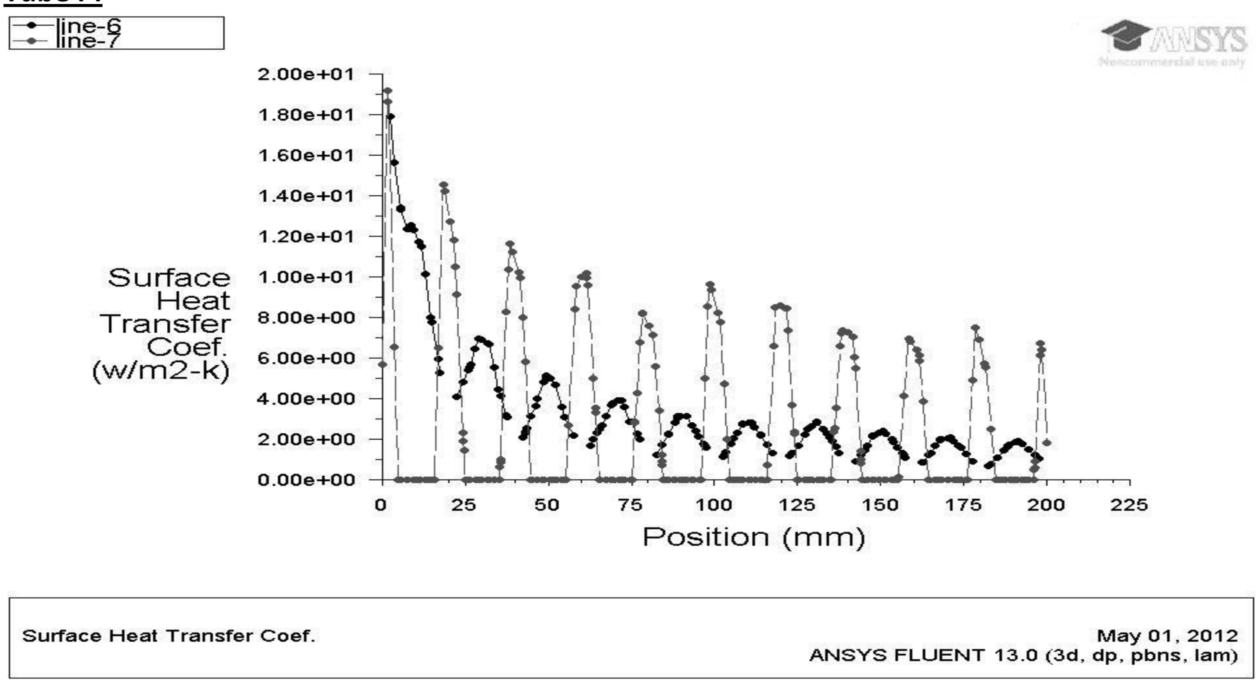


Fig. 5.20 line-6 passes from fin base and line-7 passes from fin tip.

Tube5 :

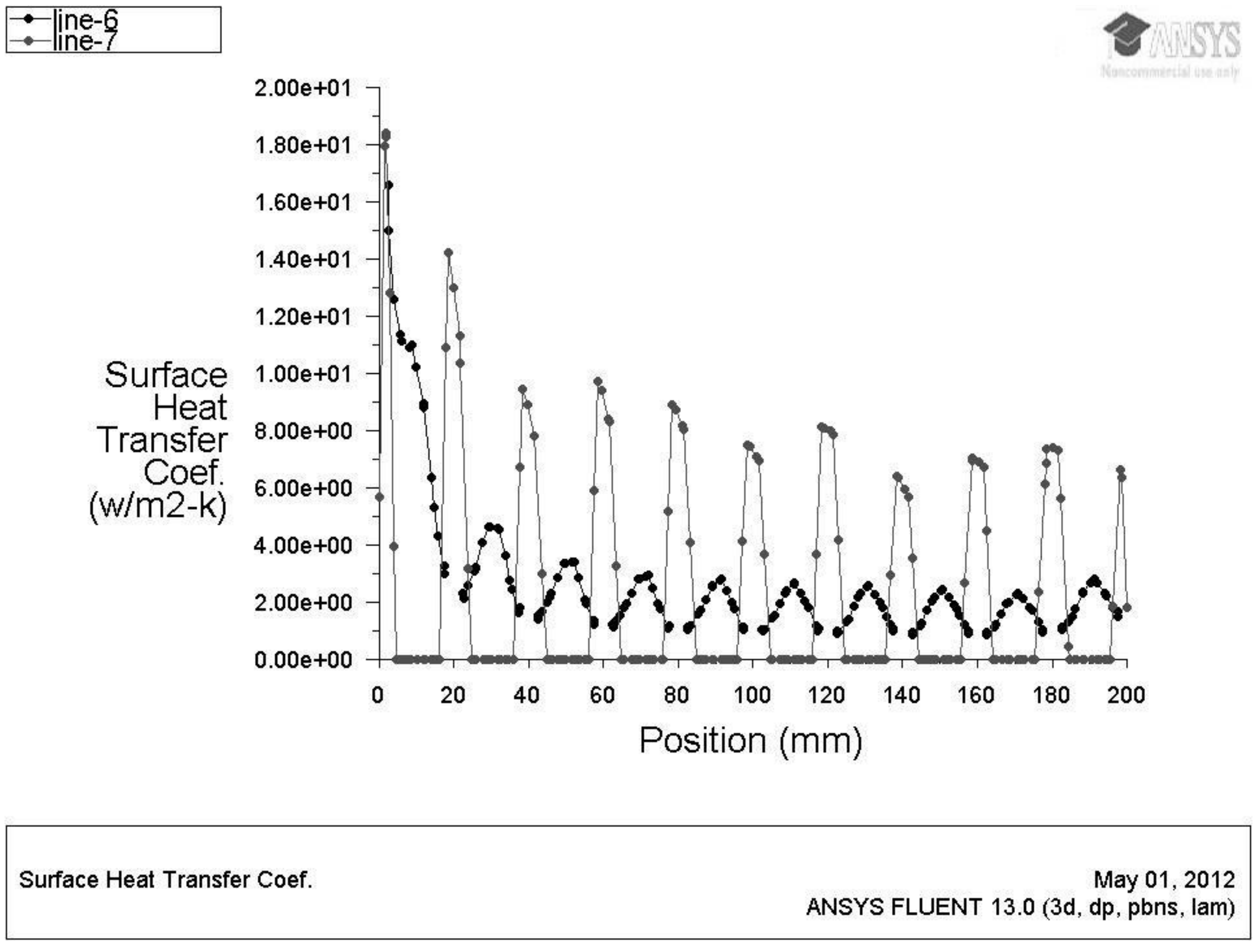


Fig. 5.21 line-6 passes from fin base and line 7 passes from fin tip.

5.4 Heat transfer rate

Tube no.	Heat transfer from fin (W)	Heat transfer from inner wall (w)
1.	---------	11.054466
2.	7.0425867	5.6044407
3.	9.6723562	7.1381139
4.	11.977043	6.2673212
5.	11.243548	5.7177154

Table 5.3 Heat transfer rate for different tubes.

CHAPTER~6

6. Conclusion

1. The tube1, tube2 and tube3 have been compared on the basis of different graphs governed from the CFD analysis and it has seen that from fig. 5.3, 5.4 and fig. 5.5, fin configuration in tube3 is more effective than other two tubes. The geometry of fins used in Tube2 has more restricted path for the air flow which increases the flow resistance and decreases the air flow rate and that downs the heat transfer rate. From fig. 5.12, fig. 5.13 and fig. 5.14, it can be seen that value of surface nusselt number has maximum value for tube3 as compared to tube1 and tube2. For tube3, near the bottom point of the tube, it is more than 300 which is greater than the tube2 which has nearly equal to 250. The surface heat transfer coefficient is compared at different position on the tube, and has more value for tube3, nearly equal to 9W/m^2-k at the lowest position of the tube, as compared to 5.5W/m^2-k and 4.5W/m^2-k for the tube1 and tube2 respectively. Heat transfer rate is 11.05 W, 12.647 W and 16.81 W respectively for the tube1, tube2 and tube3. Tube3 has maximum heat transfer rate. Hence the results showed that, for tubes having different fin configurations, the tube having ten equally spaced internal helical fins is more effective as compared to the tube without fin and tube2 which has one helical fin with large number of turns.

2. Tube3, tube4, tube5 having same fin configuration, which already had been concluded, have been compared for best fin profile. Tube3, tube4 and tube5 have rectangular, trapezoidal and concave parabolic fin profiles respectively. From fig. 5.14, fig. 5.15 and fig. 5.16, it has seen that at the position of 20mm from the bottom point of the tube the value of surface nusselt number is 450 for tube3, for tube4 it is more than 600 which is greater than tube5 which has less than 600. The value of surface heat transfer coefficient has approximately equal values for tube4 and tube5 of approximately equal to 14W/m^2-k as compared to tube3 of approximately equal to 10W/m^2-k. Heat transfer rate from tube4 is 18.244 W which is more than 17.061 W and 16.81 W for tube5 and tube3 respectively. Hence the overall performance of the fins and heat transfer rate from different fin profile has maximum value for trapezoidal fins for natural convection through internal fins for the given case.

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